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Modelling of Combustion Characteristics of a Single Curved-Cylinder Spark-Ignition Crank-Rocker Engine

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Abstract: A crank-rocker engine is a new invention used to convert oscillating motion from the curve-piston into the rotary motion of the crankshaft. The configuration of this new engine is different from the normal slider-crank engine, so the existing model used to calculate the combustion characteristic is not appropriate for this new engine. A fundamental thermodynamic model of a single curved-cylinder spark-ignition crank-rocker engine is presented. The model was simulated in MATLAB to predict the combustion characteristics at different operating conditions. The friction losses, residual gas fraction and combustion efficiency were introduced into the combustion model to improve the overall accuracy of the model. The developed model was used to analyze and evaluate the in-cylinder pressure, fuel burn rate, and heat release under various crank angle positions. To validate the predictions of the model, experimental tests were conducted on a single-cylinder crank-rocker engine at an engine speed of 2000 rpm, spark timing of 8.6° CA BTDC, full load and wide-open throttle (WOT) condition. Finally, the results were plotted and compared with the simulation results. The findings obtained from the current study have shown the ability of the simulation model to predict the combustion characteristics under different operating conditions. The agreement between the results of the present model and experimental data was reasonably good. This research work proposes a new model which can predict the behavior of the crank-rocker engine. The information gained from this study will aid in the tuning process and future development of this engine.

Keywords: thermodynamic model; combustion characteristics; fuel burn rate; heat release; crank-rocker engine

1. Introduction

Improving the internal combustion engine (ICE) performance and reducing exhaust emission are main concerns for engine manufacturers nowadays. There are more potentials to further improve engine performance and reduce emissions by utilizing an effective new engine configuration [1]. Many researchers have put efforts into developing novel engines that would be efficient and more powerful than the normal slider-crank engine. Several automotive companies have introduced new alternative engine designs, which are different from the conventional engines [2–4]. Examples of such new engines include rotary (Wankel), opposed-piston, and toroidal engines, etc.

The toroidal engine has a curved piston that oscillates inside a curved cylinder block. The idea and concept of toroidal engines have been around since the 1900s but have currently gained attention

by researchers in both academia and industry. Some technical limitations resulting from the toroidal engines have been identified as follows [5]:

- I. Leakage on the curve cylinder chamber: Due to the lack of manufacturing facilities in the past, some researchers proposed to fabricate the combustion chambers as two-piece parts. This method was not very effective due to the difficulty in sealing the combustion chamber that led to leakage and thus the decrease in the thermal efficiency of the engine. However, with the current advanced technology in additive manufacturing, a one-piece curve cylinder chamber can be easily manufactured.
- II. Engine Size: Some toroidal engines have many components, which makes it bulky and heavier, and losses occur at the pivots and joints.
- III. Higher inertial forces: With the introduction of additional elements such as rockers and piston rods, the overall inertial forces are higher than forces in the conventional engine.

Recently, several new configurations of toroidal engines employing the concept of crank-rocker (CR) mechanism have been proposed by many researchers [5–7]. The crank-rocker mechanism can become one of the most promising new engine configurations due to its simplicity in design and manufacturing.

1.1. The Crank-Rocker Engine Concept and Operating Principle

The crank-rocker engine belongs to the category of toroidal engines and it makes use of the CR mechanism concept. The main idea and concept of the crank-rocker engine shown in Figure 1a were derived from a 4-bar mechanism. It consists of four bodies (bars) connected in a close loop by four revolute joints. In this engine, the shortest bar is the crank and used as the output shaft while the rocker is used as the input bar. The connecting rod connects the input and output bars and is used to transfer the energy from the rocker to the crank link. The engine is used to convert oscillating motion from the curve-piston due to combustion force into the rotary motion of the crankshaft. The curve-piston is fixed to the rocker arm tip via bolts in order to prevent free movement of the piston (see Figure 1b).

Figure 1. Single-cylinder crank-rocker engine (a) Crank-rocker engine prototype; (b) Basic geometry of the crank-rocker engine.

It can be noted that the friction between the piston and cylinder wall of the curve cylinder can be minimized. This is because the piston is rigidly connected to the rocker arm using a bolt, and hence the piston will not deflect and slap against the cylinder wall during operation. The contact is only between the ring and the cylinder liner [7].

The purpose of using curve cylinder is to increase the turbulence intensity inside the cylinder. Hence, the curve cylinder geometry of the crank-rocker engine has a significant effect on airflow, thus resulting in better mixing, better combustion, and less emission.

The main advantage of the CR engine over most other toroidal engines is its reduced complexity and it is less bulky. In addition to that, the inertial forces for the CR engine can be dynamically balanced [8]. The engine can be modified easily to work on any types of fuels such as biodiesel, compressed natural gas, hydrogen and ethanol.

1.2. Simulation Model for ICEs

Combustion simulation models of ICEs have been widely used to evaluate engine performance and provide good assessment for engine exhaust emissions [9–13]. The simulation models allow researchers to change and test different parameters to predict engine performance under different conditions before manufacturing the real engine. The accuracy of any model depends on the model structure, availability of the information and on the assumptions made for the model.

There are many different models of frameworks used for the simulation of both compression and spark ignition engines [10,11,14,15]. The thermodynamic combustion model including heat release and physical processes of combustion are very important and helpful in investigating combustion characteristics in ICEs.

Investigating the combustion process in ICE is extremely complex because it consists of a series of complex chemical and physical processes. Several analytical and experimental techniques have been applied in IC engine in order to understand the combustion process inside the combustion chamber. With the availability of advanced research tools such as high-speed cameras, optical engines, image analysis software, and constant volume combustion chambers, many scientists and researchers have investigated combustion behaviors and processes under different operating parameters and conditions [16,17].

The in-cylinder pressure data gives a significant image and view about what is going on inside the engine [18]. In order to evaluate the combustion characteristics, it is important to take into account all engine variables such as heat transfer to the cylinder wall, changes in cylinder volume and the gas leakage (blow-by). By using the first law of thermodynamics, all the previous parameters can be calculated and analyzed. In general, there are two main approaches which have been used to quantify the combustion parameters, namely the burn rate to obtain mass fraction burned and the heat release analysis to determine absolute energy [18,19].

The most significant parameters which play important roles in the field of IC engine are the fuel burn rate and rate of heat release [20]. Many works have been carried out numerically and experimentally to investigate the effects of mass fraction burned (MFB) and heat release rate (HRR) on the engine performance and emissions over a wide range of operating conditions [21,22]. Moreover, by calculating and analyzing these parameters, both ignition delay and combustion duration can be easily identified.

The specific heat ratio is an important variable used in heat release calculations and influenced by the amount of air-fuel mixture [23]. Previous research works [24] have suggested different methods and techniques for the calculation specific heat ratio. They noted that these methods are valid and accurate for the heat release calculations.

Several approaches and techniques have been proposed by researchers [18–25] to calculate the heat release rate and mass fraction burned for both spark-ignition and compression ignition engine using single and two-zone models. Aziz et al. [26] used a single-zone model to investigate the combustion behavior of a diesel engine. They concluded that the model could predict the combustion phasing very fast and provide accurate results. Several authors made a comparison between single-zone and two-zone models using different fuels and engines [14,21,22,27–29].

From the previous findings and descriptions, it can be noted that all the modellings and analyses have been conducted on the conventional slider-crank ICEs. However, the modellings and experimental

works on toroidal engines can be rarely found in the literature. The only available information about the toroidal engines can only be found in patent documents [7].

The main aim of this work was to present a mathematical model for a CR engine. A single-zone model was developed using a MATLAB program to investigate the combustion behavior of the CR engine and to develop a better understanding of this new engine. The simulation model was used to calculate the in-cylinder pressure, heat release rate and mass fraction burned of the CR engine, and the simulation results were validated with the experimental test results. Detailed information on the development of the simulation model and experimental test data for the CR engine will be presented in the following sections.

2. Materials and Methods

2.1. The Crank-Rocker Engine Model Description

This section presents the methodology used to derive the thermodynamic model of the crank-rocker engine. The theoretical formula includes the calculation of the mechanism and kinematic properties, the formulation of thermodynamics, heat transfer and friction loss models. The model presented in this work is benchmarked against a selected model of a slider-crank engine [21] with some applicable modifications.

The model presented in this paper is almost similar to the model developed by Reference [30], but some modifications and improvements have been added to the present model such as the residual gas fraction and the kinematic properties of the CR engine which was not considered in the first model.

2.1.1. Stroke Length Calculation

A typical crank-rocker engine and the parameters that are used to define the stroke is shown in Figure 2.

Figure 2. Geometry of the CR engine.

For given link dimensions of L_1 , L_2 , L_3 , and L_4 at a certain crank position, θ_2 , the internal angles of γ , α , β , θ_3 and θ_4 can be calculated, and the equation can be summarized as follows [31].

$$BD = \sqrt{L_1^2 + L_2^2 - 2L_1L_2\cos\theta_2} \quad (1)$$

$$\gamma = \cos^{-1}\left(\frac{L_3^2 + L_4^2 - L_1^2 - L_2^2 + 2L_1L_2\cos(\theta_2)}{2L_3L_4}\right) \quad (2)$$

$$\alpha = \cos^{-1}\left(\frac{(L_1 - L_2\cos(\theta_2))}{(L_1^2 + L_2^2 - 2L_1L_2\cos(\theta_2))^{0.5}}\right) \quad (3)$$

$$\beta = \cos^{-1} \left(\frac{L_3^2 - L_4^2 + L_1^2 + L_2^2 - 2L_1L_2\cos(\theta_2)}{2L_4(L_1^2 + L_2^2 - 2L_1L_2\cos(\theta_2))^{0.5}} \right) \quad (4)$$

$$\theta_4 = 180 - \alpha - \beta \quad (5)$$

If $\theta_2 > 180$ then

$$\theta_4 = 180 + \alpha - \beta \quad (6)$$

$$\theta_3 = \tan^{-1} \left(\frac{L_4\sin(\theta_4) - L_2\sin(\theta_2)}{L_1 + L_4\cos(\theta_4) - L_2\cos(\theta_2)} \right) \quad (7)$$

As can be seen from Figure 2, the oscillation angle of the extended rocker arm between the top and bottom-dead-center positions is named as the swing angle, $\varnothing$ and is in radian. The stroke of the CR engine is unlike the conventional engine because it is not linearly dependent on the crankshaft radius. The stroke length can be changed without altering the crank radius. The crank-rocker engine stroke, S_{rocker} can be calculated by:

$$S_{rocker} = L_{41} \frac{\pi}{180} \left(2\sin^{-1} \left(\frac{L_2}{L_4} \right) \right) \quad (8)$$

where L_{41} is the length of the extended rocker arm, L_2 is the crank length and L_4 is the rocker length.

2.1.2. Piston Displacement

The piston moves in a curve-linear motion for the crank-rocker engine while in translational motion for the slider-crank engine. It can be seen from Figure 2 that as the crank rotates at a certain crank angle θ_2 , the piston moves from point x_1 to point x_2 in an arc path. The piston displacement as a function of the crank angle, x_{rocker} can be calculated as follows:

$$x_{rocker} = L_{41} \frac{\pi}{180} (180 - \partial\alpha - \partial\beta - \partial\theta_4) \quad (9)$$

2.1.3. Engine Volume

The curve cylinder is a torus and the parameters used to calculate the volume is shown in Figure 3. As the piston moves in a curve path between its extreme positions (TDC and BDC), the curved-cylinder gas volume changes and the equation can be defined as follows:

$$V(\theta) = V_c + (\pi r^2) \left(\frac{\pi \varnothing R}{180} \right) \quad (10)$$

where V is the curved-cylinder volume, V_c is the clearance volume, r is small circle radius in mm, R is big circle radius and θ is the crank angle in degree.

Figure 3. The torus cylinder geometry.

2.1.4. In-cylinder Pressure Model

The differential form of the cylinder pressure as a function of CA is obtained using the first law of thermodynamics and is defined as follows [21]:

$$\frac{dP}{d\theta} = \left(-\frac{\gamma P}{V}\right)\left(\frac{dV}{d\theta}\right) + \left(\frac{\gamma - 1}{V}\right)\left(\frac{dQ}{d\theta}\right) \quad (11)$$

where P is the cylinder pressure, γ is specific heat ratio, V is cylinder volume, and Q is the net heat input.

The heat input, $\frac{dQ}{d\theta}$ into the system can be defined by:

$$\frac{dQ}{d\theta} = Q_{in}\left(\frac{dX_b}{d\theta}\right) - \left(\frac{dQ_w}{d\theta}\right) \quad (12)$$

where Q_{in} is the heat input, X_b is the mass fraction burned, and Q_w is the rate of heat loss to the cylinder wall.

The heat generated from the engine combustion can be expressed as follows [21,23]:

$$Q_{in} = \eta_c LHV \left(\frac{1}{AF_{ac}}\right) \left(\frac{P}{R_{gc}T}\right) V_d \quad (13)$$

where η_c is the combustion efficiency, LHV is the lower heating value, AF_{ac} is the actual air to fuel ratio, R_{gc} is the gas constant, T is the mean gas temperature, V_d is displaced cylinder volume.

2.1.5. Fuel Fraction Burn Model

The fraction of fuel burned is estimated by a Wiebe function defined as [10]:

$$X_b = 1 - \exp\left[-a\left(\frac{\theta - \theta_s}{\theta_d}\right)^n\right] \quad (14)$$

where

X_b = Mass fraction burn

θ = Crank angle

θ_s = Spark timing

θ_d = Duration of combustion

a = Wiebe efficiency factor

n = Wiebe form factor

2.1.6. Heat Transfer Model

In general, the convective heat transfer through a cylinder wall can be described using Newton's law and given by [32]:

$$Q_w = hA(T - T_w)dt \quad (15)$$

where h is the coefficient heat transfer, T is the in-cylinder gas mixture temperature, T_w is the cylinder wall temperature, and A is the cylinder chamber surface area.

The coefficient of heat transfer can be defined as below [33]:

$$h = 3.26B^{-0.2}P^{0.8}T_{gas}^{-0.55}w^{0.8} \quad (16)$$

where B is the cylinder bore, P is the instantaneous cylinder pressure, T_{gas} is the average gas temperature, and w is the burned gas speed.

The burned gas speed, w defined by Woschni [34]:

$$w(\theta) = C_1 \bar{U}_p + C_2 \frac{V_d T_r}{p_r V_r} (p(\theta) - p_m) \quad (17)$$

where C_1 and C_2 are constants that vary depending on the combustion period, during the compression stroke $C_1 = 2.28$ and $C_2 = 0$, during the combustion $C_1 = 2.28$ and $C_2 = 3.24 \times 10^{-3}$, $\bar{U}_p$ is the mean piston velocity, T_r is the reference temperature, p_r is the reference pressure, V_r is the reference volume, and p_m is the motored cylinder pressure.

2.1.7. Friction Losses Model

One of the ways to improve engine fuel economy and efficiency is to reduce the mechanical friction losses. According to Monaghan, reducing the friction losses of a petrol engine by 10% could increase the fuel economy by about 5% [35]. There is a direct relationship between engine friction and engine speed. The engine friction increases quickly with the increase in engine speed, and this reduces the final power output as well as the engine efficiency. The friction means effective pressure for SI motorcycle engine has been estimated and defined by [36]:

$$FMEP = 250(S_{rocker})(N) \quad (18)$$

where N is engine speed (rev/min). The 250 is an empirical number and depending on the engine type.

2.1.8. Residual Gas Fraction Model

In all IC engines, during the exhaust stroke, not all the burned gases will move out of the cylinder. There will be some remaining gas trapped in the clearance volume, referred to as residual gas. The remaining gas will mix with the incoming air-fuel mixture during the intake stroke. The fraction of residual gas, f is defined as [11]:

$$f = \frac{1}{r_c} \left[\frac{P_i}{P_e} \right]^{\frac{1}{\gamma}} \quad (19)$$

where r_c is the compression ratio, P_i is the inlet gas pressure, P_e is the exhaust gas pressure, and γ is the specific heat ratio.

For the four-stroke OTTO gas cycle, the gas temperature at the end of the intake stroke, T_1 is defined as [11]:

$$T_1 = (1 - f)T_i + T_e f \left[1 - \left(1 - \frac{P_i}{P_e} \right) \left(\frac{\gamma - 1}{\gamma} \right) \right] \quad (20)$$

where T_i is the inlet gas temperature and P_e is the exhaust gas temperature.

To find T_1 and f , an assumption of the initial value of f is required and iteration is performed until the approximate error is less than the specified error and typical values of f are in the range of 0.03 to 0.12.

The inputs for the crank-rocker engine such as dimensions, stroke, compression ratio, number of strokes, etc. have been defined in the MATLAB script model code. Then, the piston cross-sectional area, the cylinder head area, mean piston speed, engine displacement and the clearance volume were calculated. In addition, the script code calculates the engine stroke based on the rocker extended arm, L_{41} and the swing angle, $\varnothing$. Based on the engine speed and stroke, the engine friction losses were then calculated using the equation developed by Blair [36].

In order to calculate the total cylinder volume for the crank-rocker engine as a function of the crank angle, piston displacement, length BD, transmission angle, and swing angle were calculated. All these parameters and equations have been defined earlier. Based on the MATLAB script code analysis, all the magnitudes of parameters (i.e., piston positions, length BD, transmission angle, swing angle, and total cylinder volume) of the engine were computed based on the crank angle.

The air-fuel ratio was calculated using the lambda (λ) reading, balanced equation, and stoichiometric reaction between fuel and air [30]. The general, balanced stoichiometric reaction for a gasoline fuel is defined in the code. After the previous steps, the lower heating value, atmospheric pressure, atmospheric temperature and combustion efficiency inputs were then defined.

The combustion characteristic is defined using the mass fraction burned curve. In this model, the Wiebe equation and mass of fuel inside the cylinder were defined in the MATLAB script and calculated based on the crank angle. To predict the cylinder pressure, the theory on opening and closing of intake and exhaust valves are considered in this model as suggested by Reference [21].

Upon completing the analysis of the MFB profile, the residual gas fraction inside the cylinder computed based on polytropic relationships, the corrected temperature based on ratios of the volumetric residual and inlet gases, in-cylinder pressure, heat release rate, and other engine parameters were defined on different lines and computed using the MATLAB script code. Finally, the plot statements were defined in the code based on the minimum and maximum values. The flow chart of the CR engine model description is summarized in Figure 4.

Figure 4. Flow chart of the CR engine model description.

Based on the previous formulation of the model for the combustion characteristics, MATLAB code model was developed and established for the novel crank-rocker engine. Experimental tests were conducted on the crank-rocker engine in order to validate the simulation model.

2.2. Experimental Works and Engine Set-Up

The engine used for the tests is a single-cylinder, 4-stroke, air-cooled SI engine. The engine was tested with gasoline fuel (RON 95) at full load, 2000 rpm and full throttle opening. Spark ignition was set at 8.6° CA BTDC and excess air ratio was fixed at 1.0. The crank-rocker engine specifications are listed in Table 1. The schematic diagram of the engine testbed is shown in Figure 5.

Eddy current dynamometer was attached to the engine to measure the torque output. A Kistler® spark-plug pressure transducer was used to obtain the in-cylinder pressure data using a data-logging computer named DEWETRON® which was capable of reading all the pressure data during engine operation. All components and accessories of the crank-rocker engine mounted on the test bench are shown in Figure 6.

Table 1. The specifications of the crank-rocker engine.

Parameters	Specifications
No. of Strokes	4
No. of Cylinders	1
Displacement	120 cc
Bore	55
Stroke	50.6 mm
Compression ratio	8:1
Fuel	gasoline

Figure 5. Crank-rocker gasoline engine experimental set-up. **Figure 6.** The crank-rocker engine and accessories mounted on the testbed.

Analysis of Combustion Characteristics from the Experiments

The cylinder pressure data and the corresponding crank angle position were captured via a high-speed data acquisition system. The calculations of the combustion parameters were done in an Excel spreadsheet. A special MACRO code was developed in Microsoft Excel to analyze the data. The engine input data were defined in the code. From the experimental data, the reading for the cylinder pressure and corresponding crank angle position were imported. Then, the engine displacement was calculated from the bore, stroke and number of cylinders obtained from the engine input data section. The data from the measurements of cylinder pressure were used to determine the heat release rate (HRR) and the mass fraction burn (MFB) using the equations given earlier in

this section. The code computed an average of 250 cycles and the average cycle was used to perform analysis of the combustion characteristics as a function of crank angles.

3. Uncertainty Analysis

In this work, the uncertainty analysis was conducted in order to gain a high level of confidence in the accrue of measurements. The error was analyzed and calculated based on the method suggested by Zareei et al. [37] and Holman et al. [38] which evaluates the fractional change of the measured and calculated quantities.

The probable error of each measurement can be obtained by:

$$\sigma = \sqrt{\frac{1}{N} \sum_{i=1}^N (u_i - \bar{u})^2} \quad (21)$$

where σ is the standard deviation, u_i is the quantity variables, $\bar{u}$ is the mean, and N is the number of trials.

With 95% confidence the average amount of probable error given by:

$$S_E = \pm \frac{1.96\sigma}{\sqrt{N}} \quad (22)$$

where S_E is the standard error.

3.1. Uncertainty Calculation for In-Cylinder Pressure

The uncertainty for the in-cylinder pressure was calculated using:

$$\sigma_P = \sqrt{(R_P)^2 + (S_P)^2} \quad (23)$$

where σ_P is the uncertainty in the pressure, S_P is the systematic error (from the measuring instruments), and R_P is the random error.

The systematic error for the spark plug pressure sensor was 0.20 bar, and the random error in-cylinder pressure for 250 cycles was calculated using Equations (21) and (22). Finally, the total uncertainty in the pressure is calculated as follows:

$$\sigma_P = \sqrt{(0.44)^2 + (0.20)^2} = \pm 0.48 \text{ bar}$$

3.2. Mean Absolute Percentage Error (MAPE)

The mean absolute percentage error is the most common statistical measure used to estimate the accuracy of the model as a percentage. The error percentage between the simulations and experiments was calculated using the following formula [39]:

$$MAPE = \frac{1}{n} \sum \left| \frac{(E - S)}{E} \right| * 100 \quad (24)$$

where n is the number of data, E is the experimental value, and S is the simulation data value.

4. Results and Discussion

The simulated results from MATLAB model were compared with the corresponding experimental data. The combustion characteristics are expressed in term of cylinder pressure, HRR, P-V diagram and MFB. The experimental results were compared with the simulation results at an engine speed of 2000 rpm and full load condition.

Figure 7 shows the simulated pressure compared with the experimental pressure measured from the crank-rocker engine at 2000 rpm. In general, the simulated pressure curve shows a very good agreement and almost matches with the experimental pressure. The reason is due to the formulation and accuracy of the model that includes the friction losses model, gas residual model, intake valve opening and exhaust valve closing operation model. As can be seen from Figure 7, the simulated pressure values are less than the experimental values and this can be obviously seen in the compression stroke. The reason is due to the assumptions made in the simulation model where it was assumed that the specific heat ratio is the same for all the engine strokes, i.e., 1.35. In reality, the specific heat ratio varies with the temperatures. The other reasons are due to the assumptions made for the initial conditions such as atmospheric pressure, temperature, and specific heat capacity values which are clearly dependent on the temperature. The large difference in temperature can result in significant errors. It can be noted that the maximum relative error between the simulated and experimental pressure was found to be about 17.54%.

Figure 7. In-cylinder pressure vs. crank angle for the experiments (solid red line) and simulations (dash blue line).

In order to understand the work produced by the crank-rocker engine, the pressure-volume profile was plotted and analyzed. The pressure-volume diagram, which was obtained from the experimental data at 2000 rpm, is shown in Figure 8. The pressure-volume diagram from the simulation results was also plotted on the same figure for comparison purposes. It is remarkable to note that the P-V diagram shape from the simulation model had a similar trend shape as the experimental result.

Comparisons between experimental and simulated results of mass fraction burned (MFB) as a function of the crank angle are shown in Figure 9. The figure demonstrates that the predicted mass fraction burned curve matches well with experimental data. In addition, it can be concluded from the figure that the Weibe function matches the characteristics of an 'S' curve.

The heat release rate (HRR) at WOT condition and 2000 rpm as a function of the crank angle has been simulated and plotted along with experimental results shown in Figure 10. Despite the fact that the trend curve between the simulation and experiential data matched with each other and the relative error was about 19.85%, there was some variation in the readings, especially during the intake stroke. This is possibly due to the assumptions made for the simulation model such as initial atmospheric conditions and specific heat capacity values.

Figure 8. In-cylinder pressure vs. volume for the experiments (solid red line) and simulations (dash blue line).

Figure 9. Mass fraction burned vs. crank angle for the experiments (solid red line) and simulations (dash blue line).

Figure 10. Heat release rate vs. crank angle for the experiments (solid red line) and simulations (dash blue line).

5. Conclusions

This paper has presented a mathematical model for theoretical predictions of combustion characteristics of a CR engine. The simulation results were compared with the experimental results for comparison and validation purposes. The following conclusions were made as to the outcome of the research work:

1. It has been observed that the results from the simulation model, i.e., volume, mass fraction, pressure and pressure-volume, and heat release rate followed the expected characteristics. The simulations results show a good correlation with the experimental results.
2. There was a relative error due to the assumption made in the simulation model such as the specific heat ratio and specific heat capacity values which were clearly dependent on the temperature. The large difference in temperature can result in significant errors.
3. The model predicts changes in in-cylinder pressure, P-V, MFB, and HRR.
4. The thermodynamic simulation model can be further improved by taking into account all mechanical losses and selecting the right parameters.

The basic fundamental theory in this present work is to investigate the combustion behavior of the new crank-rocker engine and to predict its performances. The information gained from this study will aid in the fine-tuning process and future development of this engine. This will also create opportunities for new researchers to be involved in the design studies of the CR engine which may become a substitute for the current conventional engine technology.

6. Patents

This work has been patented under the number MY-114703-A.

Author Contributions: All authors have contributed to the current work. S.E.M. performed the experiments test, built the mathematical model, analyzed the data and write the first draft of the paper; M.B.B. came up with the research idea, contributed to the paper revision and analyzed the data; A.R.A.A. contributed to the reviewing of the manuscript; and E.Z.Z.A. contributed to the analysis of results.

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